N74-10244

PRECEDING PAGE BLANK NOT FILMED

Paper No. 13

COMPARATIVE EVALUATION OF PREDICTED AND MEASURED PERFORMANCE OF A 68m³ TRUNCATED REVERBERANT NOISE CHAMBER

H. D. Cyphers, A. N. Munson and F. J. On; *Goddard Space Flight Center, Greenbelt, Maryland*

ABSTRACT

The performance of a medium size, truncated reverberation chamber is evaluated in detail. Chamber performance parameters are predicted, using classical acoustic theory, and compared with results from actual chamber measurements. Discrepancies are discussed in relation to several available empirical corrections developed by other researchers. Of more practical interest is the confirmation of a recent theory stating that the present rule of thumb for the ratio of specimen volume to test chamber volume, approximately ten percent, is overly conservative and can be increased by a factor of at least two and possibly three. Results and theoretical justification of these findings are presented.

INTRODUCTION

This paper details the results of an evaluation of the acoustic field produced within a new reverberation chamber constructed by the Structural Dynamics Branch at Goddard Space Flight Center. The chamber, with a total volume of $68m^3$ (2,400 ft.^{3}), was constructed to provide a means of experimentally verifying the results of theoretical studies in the areas of noise field prediction and the effects of various sized test volumes on the acoustic field. In addition, the chamber is frequently used to test small scientific spacecraft and experiments.

At the completion of the empty evaluation and prior to beginning a series of tests using various test volume sizes, a requirement to acoustically test the German Helios spacecraft developed. This spacecraft, with an enclosed volume of approximately $14 m^3$ or 20% of the chamber volume, afforded an opportunity to obtain additional data on the effects of exceeding the 10:1 rule-of-thumb ratio for chamber to test volume. The results of this test are discussed in relation to the empty chamber performance.

The development of a criteria for acceptable performance of a room as a reverberant acoustic testing facility requires several considerations. For a given room design and acoustic driver, the first important consideration is the modal density (or the modal overlap index) of the room which indicates the frequency distribution of the acoustic modes in the room. The second consideration is the spatial variations in the sound pressure levels which indicate the degree of uniformity (or nonuniformity) of the sound field in the room. Other considerations are the maximum achievable sound pressure level and the spectra shaping capability of the reverberant room.

The performance of a well designed reverberation chamber depends strongly on the modal overlap index. In the frequency regime where the modal overlap index is very much less than one, the chamber acoustic field is usually unpredictable and the spatial variations are large. In the frequency regime where the modal overlap index is of the order unity or larger, the chamber acoustic field is well defined in a statistical sense and the spatial variations are relatively small. These considerations dictate the lowest usable frequency that the chamber can be used for reverberant acoustic testing.

Accordingly, several descriptors are commonly used to specify the performance of reverberation chambers. The most important of these are: the lowest usable test frequency and the ability to match the required spectra shapes and levels of vehicle specifications. In evaluating these characteristics, the following properties of the reverberation room were determined: reverberation time (or effective absorption), modal density (or modal overlap) and spatial variation of the sound pressure field. Each of these areas will be addressed in detail in the Test Results and Discussion section of this paper.

DESCRIPTION OF TEST FACILITY/SETUP

Figure 1 is an isometric view of the reverberation noise test facility. The chamber is rectangular in shape with two walls bisected in the vertical plane to conform to the space available. The practicability of this design was verified by a series of modal studies (Reference 1). The maximum orthogonal dimensions are in the ratio 1.00:0.79:0.63 suggested by Sepmeyer (Reference 2) for good chamber performance.

The acoustic excitation to the reverberant chamber is supplied by a 60 Hz exponential horn which can be driven by either a Noraircoustic Mark V (50 Kw) or a Ling EPT-94B (4 Kw) air modulated acoustic driver.

The control and data collection system consists of standard, laboratory-quality equipment. Spectrum shaping and control is done on a one-third octave basis in real time. A multi-channel microphone averager[1] is used to assure that the control signal is a good representation of the noise field.

All offline data analyses were done using the power spectral density, one-third octave band and coherence algorithms of the Goddard dynamic data-analysis computer program (DYVAN). This program is designed to handle sinusoidal-sweep, shock transient, or stationary random data.

In the experimental evaluation of the chamber performance descriptors, careful selection of microphone locations was required to accurately measure the acoustic field in the chamber. As the sound pressure levels in the chamber are higher nearer the walls, corners and edges than throughout the rest of the room, all microphones were placed at least $3/4\lambda$ away from the corners and edges of the chamber and at least $\lambda/4$ from the walls where λ is the wavelength of the lowest frequency of interest.

In addition to observing boundary conditions in locating microphones, the positions of the microphones relative to each other was selected, in the case of the empty chamber evaluation, to provide maximum

---

[1] The use of multi-channel averaging control insures that the control sound pressure level will approach the average SPL over the test volume; i.e., the control SPL is centered within the inherent spatial variation of the chamber.

statistical independence (coherence approaching zero). This is desired in evaluating spatial variation of the noise field.

For the occupied chamber data, the microphone positions specified by the Helios test procedure were used. Boundary conditions were not satisfied as in the empty chamber evaluation due to the size of the spacecraft. Figure 2 shows the Helios spacecraft assembled in the chamber. Figure 3 shows the locations of the control and monitoring microphones.

The worst case coherence results, based on data from the empty and occupied chamber acoustic tests, are presented in Figure 4 to indicate the approximate degree of statistical independence of the microphone outputs actually achieved via the use of these microphone locations.

The theoretical plot of Figure 4 was computed from

$$C_{AB\Delta f} = \left[\frac{\sin \frac{2\pi}{c} fr}{\frac{2\pi}{c} fr} \right]^2 \qquad (1)$$

where r is distance between microphone locations, f is center frequency and c is speed of sound. In practice it is reasonable to use this equation as a standard for determining the degree of statistical independence between microphone outputs.

TEST RESULTS AND DISCUSSION

The results of the study will be categorized in terms of specific properties of the acoustic field within the reverberation chamber. Each property will be discussed briefly from a theoretical standpoint; the predicted and measured results for the chamber will then be given along with a discussion of the method used and any unusual features of the data. The operational or performance characteristics of the reverberant acoustic facility, as derived from the measured properties, will then be summarized and discussed briefly from both a theoretical and applications point of view.

The necessity of obtaining chamber performance parameters during the actual Helios test limited the ability to measure all of the desired properties. The results of adding a 20% test volume will be discussed under the section for which it was possible to obtain measured data.

The following properties of the room were determined: the reverberation time (or effective absorption), modal density (or modal overlap) and spatial variation. The most important operational characteristics from a user's standpoint are: the lowest usable frequency and the ability to match the required spectra shapes and levels of vehicle specifications.

Reverberation Time (Effective Absorption)

The reverberation time for a given frequency is the time required for the average sound pressure level, originally in a steady state, to decrease 60dB after the source has ceased. This decay is caused by absorption of the sound energy not only at the boundaries of a room, but also in the air itself.

The reverberation time is related to the total effective absorption of the reverberant room by Sabine's equation (Reference 3):

$$T_{60} = \frac{60V}{1.085c(a+4mV)} . \qquad (2)$$

where T_{60} = reverberation time in seconds

V = volume of room in cubic meters

c = speed of sound in meters per second

a = number of metric absorption units in square meters

m = energy attenuation constant in meters $\times 10^{-1}$

The term, $a+4mV$, is the total effective absorption where $\underline{a}$ is the absorption at the boundaries (walls) and $\underline{m}$ is the energy attenuation constant in the transmitting medium.

In the chamber evaluated, predicted effective absorption losses were computed utilizing model study data.

Figure 5 is a plot of the predicted and measured reverberation times. The times plotted are the average of three measurements taken at elevations of 1.50m (5 ft.), 3.04m (10 ft.) and 4.12m (13.5 ft.) on a vertical line through the center of the chamber. Also plotted are the predicted and actual effective absorption coefficients. The actual effective absorption was calculated from Equation 2.

The measured reverberation times possess some scatter and are sometimes multivalued below approximately 400 Hz. Above 400 Hz, the reverberation times decrease smoothly with increasing frequency. The scatter at the lower frequencies is due to expected sound pressure fluctuations in an enclosure of this size. Morse and Ingard (Reference 4) discuss in detail the reasons for the multiple slope and pressure fluctuation phenomena observed in some of the reverberation data.

Several means of exciting the chamber were tried to determine the effects of various types of excitation on the reverberation time. Figure 6 is a plot of reverberation times obtained using wide band white noise and discrete frequency excitation. As expected, the discrete frequency excitation values differ significantly, especially below 400 Hz, from random noise excitation because fewer standing waves are excited at a given time. The wide band white noise excitation (with the microphone response filtered on a one-third octave basis) agrees quite closely with the previous data. Although other experimenters seem to prefer one-third octave band excitations, wide band white noise excitation can provide representative data at considerably less effort and expense.

In the scaled model study (Reference 1), absorption coefficients were calculated from actual reverberation times for both the truncated and equivalent rectangular (dimensions equal to parallel dimensions of truncated chamber) model chambers. The absorption coefficients for the two configurations were approximately the same at all frequencies. Since the above data (Figure 5) show that the calculated absorption coefficient for the Goddard $68m^3$ chamber agrees with that predicted using the scaled model study results, it may be assumed that the reverberation times will bear the same relationship as well. That is, the predicted reverberation time of the truncated chamber is approximately 85% of the reverberation time of an "equivalent" rectangular chamber in accordance with the dependency of the reverberation time on the volume to area ratio of the two chambers. No reverberation time data was obtained for the occupied chamber condition.

Modal Density, Sine Sweep Response, Modal Overlap

The frequency response of a reverberation room is characterized by a series of discrete resonant modes. For a simple, normal rectangular room, the lowest or first of these resonant modal frequencies occurs when one-half the acoustic wave length equals the longest dimension of the room. The second resonant frequency occurs when one-half the wavelength equals the next longest dimension of the room. An entire harmonic series containing integer multiples of the fundamental axial modes exists for each of the three room axes.

At frequencies higher than those of the fundamental axial modes, more complex modes appear. These include modes associated with four walls in which the standing waves are tangential to one pair of walls, and oblique modes, which involve the interaction of standing waves between all six walls.

The aim in reverberation chamber design is to achieve a sound field within the volume which is completely diffuse in all frequency ranges of interest. This implies that there are many modes uniformly spaced in frequency and that all angles of incidence of the sound are equally probable.

The modal frequencies of a rectangular chamber can be calculated from the characteristic equation given by (Reference 5)

$$f_{m,n,p} = \frac{c}{2} \left[\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2 + \left(\frac{p}{L_z}\right)^2 \right]^{\frac{1}{2}} \qquad (3)$$

where _c_ is the speed of sound in air (meters/second at 20°C) and L_x, L_y, and L_z are the room dimensions in meters, _m_, _n_ and _p_ are mode numbers for the X, Y and Z modes respectively and take on the integer values 0, 1, 2, 3, ... etc.

Near the fundamental frequency of the room, the effective absorption coefficient is very dependent on the number of modes, thus contributing to the nonuniformity of sound level with frequency. As an approximation, Equation 3 was used in estimating the modal frequencies of the Goddard truncated chamber. The actual data indicate that many more modes are actually excited. Extensive computation of the modal frequencies of reverberation rooms using the classical Equation 3 is tedious unless handled by computers. It is generally more significant to know the approximate number of acoustic modes that might be excited within any given bandwidth Δf (i.e., the product of modal density and bandwidth).

Modal Density [n(f)]--Modal density is the frequency distribution of the acoustic modes (the number of modes per frequency bandwidth as the bandwidth approaches zero) in the room due to the excitation of the room by a sound source. It is largely dependent on the physical dimensions of the enclosure as well as the excitation frequency.

The asymptotic modal density of a rectangular reverberation room may be computed by Equation 4 which is derived from Equation 3 (Reference 6)

$$n(f) = \frac{4\pi f^2 V}{C^3} + \frac{\pi f A}{2C^2} + \frac{P}{8C} \qquad (4)$$

where _P_ is the length of chamber wall intersections (total length of all chamber edges), _V_ is volume in cubic meters, _A_ is surface area in square meters, _C_ is velocity of sound in meters/second and _f_ is frequency. This equation has also been shown to be applicable to this particular truncated chamber shape (Reference 1). Accordingly, the number of resonant modes of the truncated chamber in each band of frequencies Δf may be computed by

$$N_{\Delta f}(f) = n(f)\ \Delta f = \left(\frac{4\pi f^2 V}{C^3} + \frac{\pi f A}{2C^2} + \frac{P}{8C}\right) \Delta f \quad (5)$$

where f is center frequency of the band.

Alternatively, the cumulative resonant frequency distribution (i.e., the integral of the modal density)

$$N(f) = \int_0^f n(f)df = \frac{4}{3}\ \frac{\pi f^3 V}{C^3} + \frac{\pi f^2 A}{4C^2} + \frac{Pf}{8C} \quad (6)$$

may be determined for comparison with a cumulative count of the sine sweep peaks obtained by counting the number of modes excited during a sinusoidal sweep survey of the chamber.

Sine Sweep Response--In making sine sweeps, the sweep rate and direction of sweep, up or down, can influence the number of resonances excited. A sweep rate of 2 minutes per octave gave good modal definition and repeatability when sweeping from either direction.

Figure 7 is a plot of the predicted modal density [n(f)], predicted cumulative resonant frequency distribution [N(f)] and the predicted and measured mode counts ($N_{\Delta f}$) for each one-third octave band in the chamber. The criterion used for counting the sine sweep response peaks required that the peak-to-valley ratio, on both sides of the peak, be 3dB or more in order that the peak be counted.

The results presented in Figure 7 indicate that, for the chamber under study, Equation 5 provides a conservative prediction of the number of modes on a one-third octave band basis. Above approximately 250 Hz the resonance peak count method greatly underestimates the number of modes that actually exist. This deviation from the predicted distribution is attributed to: (1) peaks that are missed in the counting procedure when the response peaks are closely spaced in frequency, and (2) many response modes occurring within the bandwidth of a single mode. In these cases, counting the response peaks does not yield an accurate measure of the actual number of modal resonances (Reference 7). The first cause can be minimized by using a higher paper speed to expand the frequency scale.

Also shown on Figure 7 is the measured one-third-octave band mode counts of the occupied chamber. The number of modes is seen to be considerably lower than in the empty chamber condition. This is due primarily to the way the data was taken. The data was obtained by counting the peaks in a power spectral density plot of the microphone response using a 2 Hz analysis bandwidth. The peak to valley mode height criterion used during the empty chamber sine sweep evaluation was reduced to ±1dB to account for the averaging effect over the 2 Hz bandwidth. Because of the relatively broad analysis bandwidth used, this method of counting modes is very conservative. Based on a criteria of a minimum of 7 modes per one-third-octave bandwidth, the chamber is usable (considering only modal density) down to 200 Hz.

Modal Overlap (M)--The modal overlap index provides a measure of the diffuseness of the noise field in a well-designed reverberation room. It is defined as the number of adjacent modes which lie inside the 3dB bandwidth of a typical chamber modal resonance. Computationally, it is the product of the 3dB bandwidth $\Delta(f)$, and the average modal density, $n(f)$:

$$M(f) = \Delta(f)\ n(f) \qquad (7)$$

where

$$\Delta(f) = 2.2/T_{60} \qquad (8)$$

The 3dB bandwidth of a typical chamber modal resonance, $\Delta(f)$, is defined by the chamber resonant Q determined from the reverberation time. It will be recalled that one-third octave band noise excitation was used in this study to establish reverberation times. The modal density, $n(f)$, as shown in Figure 7, is the average modal density for the one-third octave band of interest with center frequency f.

The product of the 3dB bandwidth, $\Delta(f)$, and the average modal density, $n(f)$, assumed constant over the bandwidth used when counting the modes, yields a figure of merit in terms of modes per chamber resonant bandwidth. This term, together with the spatial variation of sound pressure levels in the chamber, provides a means of specifying the lowest usable frequency of the chamber. Selection of an appropriate value of M (f) is considered further in the section on determining the lowest usable frequency. Figure 8 shows the predicted and measured values of modal overlap. Above 250 Hz it was necessary to use predicted values for the modal overlap due to the difficulty in measuring the modal density by the peak response count method.

The modal overlap was not computed for the occupied chamber because of the lack of measured reverberation data.

Spatial Variation (S)

The degree of spatial nonuniformity (commonly termed spatial variation) of the acoustic noise levels in the reverberant chamber was determined in terms of the standard deviation, S, of sound pressure level. The deviation was computed from a sufficiently large set of uncorrelated sound pressure levels measured at specified locations in the chamber with a specific noise excitation. In Figure 8, the predicted and measured results of spatial variation for the empty and occupied reverberant chamber are presented. The empty chamber modal overlap has also been plotted in this figure to facilitate the application of the criteria for determining the lowest usable frequency of the chamber.

The measured empty chamber spatial variation results were determined from a set of seventy-two sound pressure levels, L_i, measured at specified locations. In terms of standard deviation, S, the spatial variation was computed from

$$S = \left[\frac{n \sum_{i=1}^{n} L_i^2 - \left(\sum_{i=1}^{n} L_i \right)^2}{n(n-1)} \right]^{\frac{1}{2}} \qquad (9)$$

where n=72 is the total number of measurements in the set. This determination was repeated for each one-third octave band and is independent of the mean level, (i.e., S would not change if the experiments were repeated at a different sound pressure level), as well as independent of time since the individual sample levels, L_i, were already time-averaged quantities computed via the DYVAN program.

The spatial variation data for the occupied chamber was calculated from the seven microphones (n=7) used for control and monitoring during the Helios test. The statistical independence of those microphones at these locations is illustrated in Figure 4. As noted previously, some of these microphones were located within inches of the spacecraft skin and thus were subject to some boundary effects.

The spatial variation data presented in Figure 8 for the empty chamber condition indicates a deviation of approximately 0.5dB between predicted and measured values at frequencies above 500 Hz corresponding to modal overlap indices greater than 1.8. Measured values of spatial variation exceed ±3dB at frequencies below 160 Hz.

The spatial variation for the occupied chamber yielded the same general shape as the empty chamber results. The noted irregularities were probably due to the boundary effects. In this case, the frequency above which the measured values of spatial variation exceed ±3dB has shifted upward less than one octave to approximately 260 Hz.

Lowest Usable Frequency (f_c)

The lowest usable frequency for a reverberation chamber (not to be confused with horn coupling cutoff frequency) is that frequency below which the sound field in the reverberation room ceases to have an acceptable level of spatial variation and diffuseness. This frequency may be affected by the size of the test specimen occupying the chamber.

Based on the criteria that for one-third octave band sound pressures, the lowest one-third band showing no more than ±3dB spatial variations and no less than one-third modal overlap index shall be specified as the lowest acceptable test frequency, the data presented in Figure 8 thus indicate that the chamber is useful at frequencies above 160 Hz for test volumes less than 10 percent.

In the case of the chamber occupied by a test specimen of about 20%, the lowest usable frequency of the chamber cannot be established via the criteria on modal overlap index since the index cannot be determined without the use of reverberation times. Employing an equivalent criteria based on modal density consideration, together with the criteria on spatial variation, the lowest usable frequency for the 20% volume specimen occupied chamber case shifted to 260 Hz.

In terms of the modal overlap index M, known volume V in cubic meters and the reverberation time T_{60} in seconds for the mode in question, the lowest usable frequency may also be computed by:

$$f_c = \left[\frac{Mc^3 T_{60}}{8.8\pi V}\right]^{\frac{1}{2}} \qquad (10)$$

The value for M corresponds to the minimum value of Modal Overlap Index ($1/3 \leq M \leq 3$) selected on the basis of an acceptable level of spatial variations of sound pressure levels within the chamber. Recall that M was based on the chamber resonant bandwidth. Equation 10 can be derived using the first term of Equations 5 and 7.

The underlying logic for establishing the above criteria on the basis of a modal density or modal overlap requirement is as follows: For qualification testing or design evaluation, it is frequently desirable to express the lowest usable frequency in terms of the number of modes required within the resonant bandwidth of a structure. The required modal density can be directly related to the anticipated Q of the test specimen and the average number of modes desired within the specimen's resonant band to give the proper response simulation by the following relationship:

$$n(f)\ \Delta f_r \simeq \frac{4\pi V f_o^{\,3}}{C^3 Q} \geq \text{minimum desired or specified number of modes} \qquad (11)$$

522-137 O - 73 - 11

where $\Delta f_r = f_o/Q$

f_o = fundamental structural mode frequency

Q = resonant amplification factor

Example: First establish the number of modes required within a resonant bandwidth of a structure. If a minimum of 7 is required, then the minimum structural resonant frequency, f_o, which can be determined. Using the following values in the equation:

$V = 68m^3$ (2,400 ft.^{3})

C = 344m/sec. (1,128 ft./sec.)

Q = 10 (expected maximum value)

f_o is calculated to be 150 Hz. For a resonant frequency of 150 Hz and an expected Q of 10 we have a resonant bandwidth of 15 Hz within which a minimum of seven chamber resonant modes are required. Figure 7 gives a plot of the actual number of modes, n(f), at a given frequency for empty chamber conditions. Entering this graph at 150 Hz, one can expect to have at least 0.7 modes at this frequency. Multiplying the number of chamber modes (0.7) and the structural resonant bandwidth (15 Hz) yields 10.5 modes in the lowest frequency band of interest (Δf_r). Note that if this criteria is fulfilled at 150 Hz, it will be exceeded, on the average, above 150 Hz, since the modal density increases with frequency. Therefore, it appears reasonable to define a lower bound frequency for a reverberant room as the lowest frequency at which the room has the required modal density, provided the spatial variation requirement is not exceeded.

Spectra Shaping Capability

The ability of a generator/chamber configuration to reproduce a given noise spectra is one of the most important requirements to be satisfied by the designer. For the $68m^3$ chamber, the reproducibility of noise spectra for three commonly used launch vehicles (Atlas Centaur, Improved Delta and Titan III-C) and for the acoustic noise portion of DOD Military Standard 810B was assessed. In all cases, the noise spectra was reproducible within ±3dB.

In testing the Helios to the Titan III-C spectrum, insignificant differences were observed between the four channel average of the empty chamber spectrum and the spectrum obtained with the Helios in the chamber. On the basis of these results, a test specimen volume of 20% of the chamber volume indicates negligible effect on the spectral shape as normally measured during acoustic tests. Wyle Laboratories has done some model study work in this area which supports these tentative conclusions (Reference 8).

Figure 9 is a plot taken from the Wyle study. This data was taken from a relatively small scale model with room dimensions ratios of 1:.82:.71. The plot shows the sound pressure levels measured in the chamber with various specimen sizes relative to sound pressure levels measured in the empty room. Superimposed on this plot are similar results from the Helios test.

The results from the GSFC full-sized chamber are more promising than was expected from the scale model data. The increasing difference in SPL between the occupied and empty chamber results, with increasing specimen size, at the low frequency end is primarily due to reduction in chamber volume caused by the specimen. An extended frequency scale employed in the model study to collect sine sweep test data shows that the longer average path lengths followed by the sound rays in passing around the specimen results in a general lowering of the individual resonances of the chamber. This lowering of the resonances has the effect of spreading the peaks out in the lower frequency range so that the chamber is even less usable at these low frequencies than when it is empty. This spreading out of the peaks was also demonstrated by the increase of spatial variation in Figure 8. Based on these results, the GSFC chamber occupied by a specimen having a volume of 20% of the chamber volume would not be considered to seriously degrade the chamber performance. Moreover, for specimens having sufficiently low resonant magnification factors, a specimen as large as 50% of chamber volume possibly could be tested.

CONCLUSIONS

The purpose of this investigation was to obtain experimental data with which to verify the predicted performance of the truncated reverberation chamber and to assess the effect of various sized test volumes on chamber performance. From the results of the study, the following conclusions can be drawn:

1. The extremely close correlation obtained between predicted, model study, and measured data permit using existing theory to predict the chamber noise field and test item response with a high degree of confidence.

2. Test specimen volumes of up to 20% of the chamber volume are shown to have a negligible effect on the acoustic noise field when measured on an averaged one-third-octave basis. A closer examination on a narrow band statistical basis reveals only small changes which, when understood by the test engineer, give him a basis for justifying the testing of even larger test volumes of up to 50% of the chamber volume.

3. A simplified method of measuring reverberation time using broadband rather than the conventional narrow band noise excitation yielded, for all practical purposes, identical results.

4. All presently used and anticipated acoustic test specifications can be satisfactorily reproduced. The lowest usable frequency, assuming a maximum allowable spatial variation of ±3.0dB is 160 Hz. For many tests, the chamber is adequate below this frequency. Data are presented for determining the degree of compromise necessary in chamber performance for operation at lower frequencies.

5. Maximum versatility in spectra shaping is obtained by using two air modulators. While the bulk of this evaluation was done using

the Noraircoustic Mark V generator, a limited amount was done using the lower powered Ling EPT-94B as a noise source. From extensive previous experience with the Ling source, the authors feel that this generator will provide superior shaping capability (at lower overall levels, 148dB maximum) on a one-third octave band basis over the range of 100-1,000 Hz.

Due to space limitations, considerable background data, theoretical justifications, and discussions on predicting overall and spectral sound pressure levels have been omitted in this paper. A NASA technical note covering these areas is presently being published.

REFERENCES

1. Scharton, T. D., Rentz, P. E., Lubman, D., and White, P. H., "Techniques for Improving the Low Frequency Performance of Small Reverberation Chambers." A report prepared for the National Aeronautics and Space Administration, NAS5-21003. California: Bolt, Beranek and Newman, Inc., (1970).

2. Sepmeyer, L. W., "Computer Frequency and Angular Distribution of the Normal Modes of Vibration in Rectangular Rooms," J. Acoust. Soc. Am. 37, 3, (1965).

3. Beranek, L. L., "Acoustics" McGraw-Hill Book Company, Inc., New York (1954).

4. Morse, P. M., and Ingard, K. U., "Theoretical Acoustics," McGraw-Hill Book Company, Inc., New York, Pg. 597, (1968).

5. Beranek, L. L., "Noise Reduction," McGraw-Hill Book Company, Inc., New York, Pg. 235, (1948).

6. Morse, P. M., "Vibration and Sound," McGraw-Hill Book Company, Inc., New York, Section 32 (1948).

7. Mills, J. F., "A Study of Reverberation Chamber Characteristics," Institute of Environmental Sciences, Proceedings, Fig. 10 (1967).

8. Cockburn, J. A., "Evaluation of Acoustic Testing Techniques for Spacecraft Systems." A report prepared for the National Aeronautics and Space Administration, NAS5-21203. Alabama: Wyle Laboratories (1971).

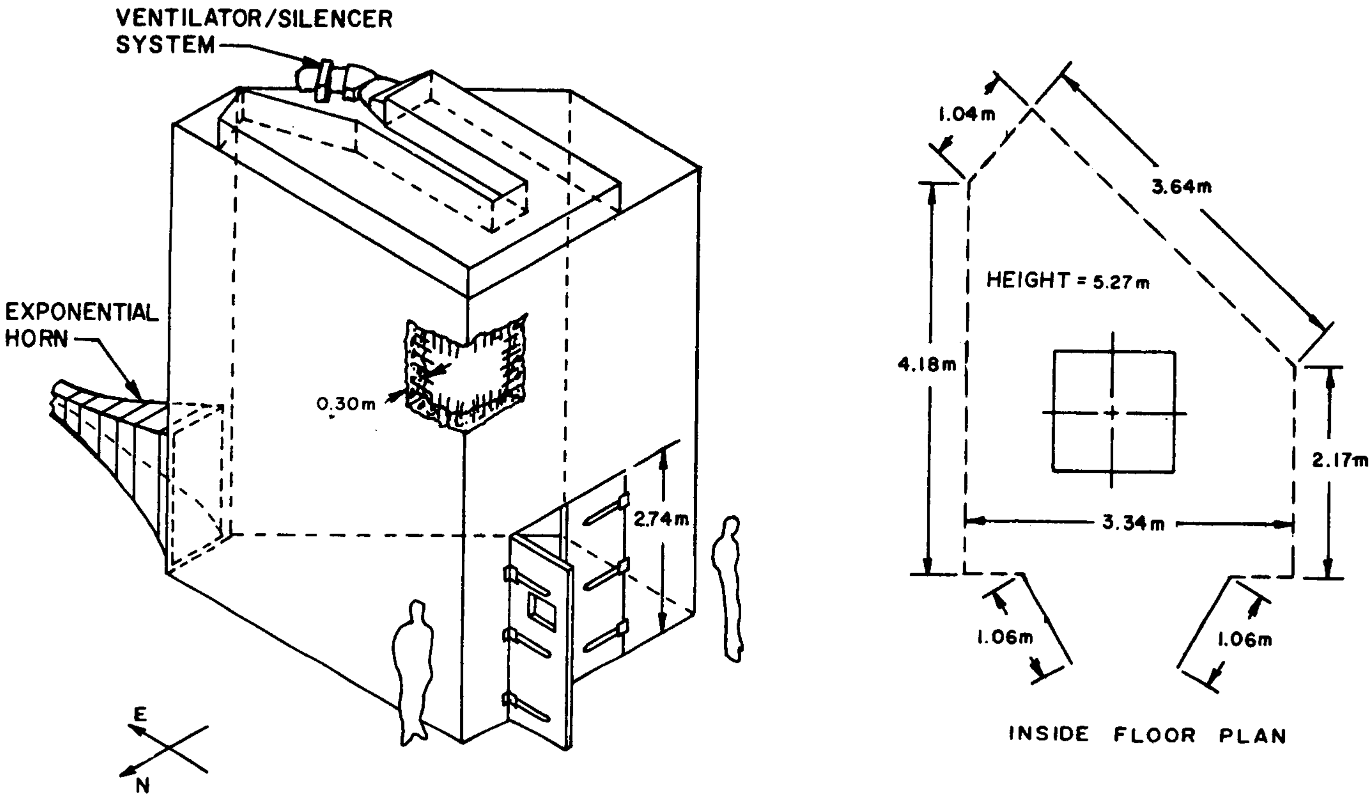

FIGURE 1. Goddard Reverberant Noise Chamber

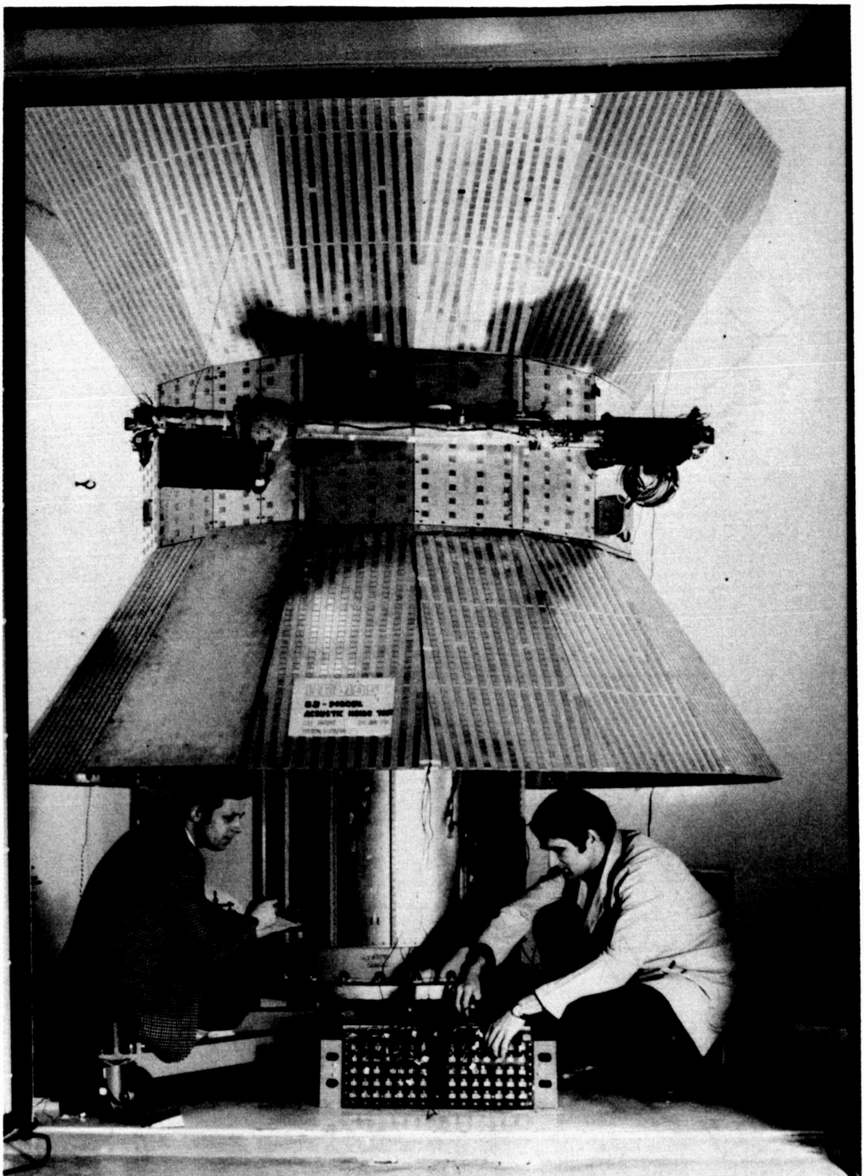

Figure 2. Helios Spacecraft Mounted in $68m^3$ Chamber

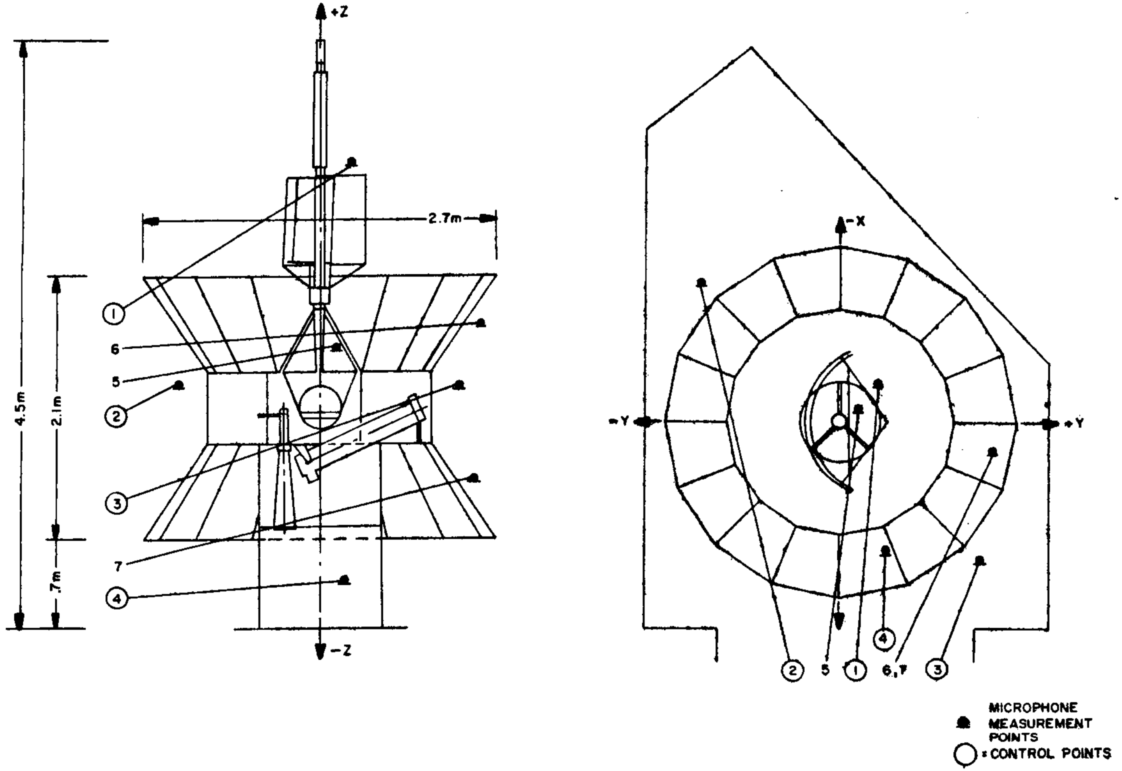

FIGURE 3. HELIOS SPACECRAFT SHOWING MICROPHONE LOCATIONS

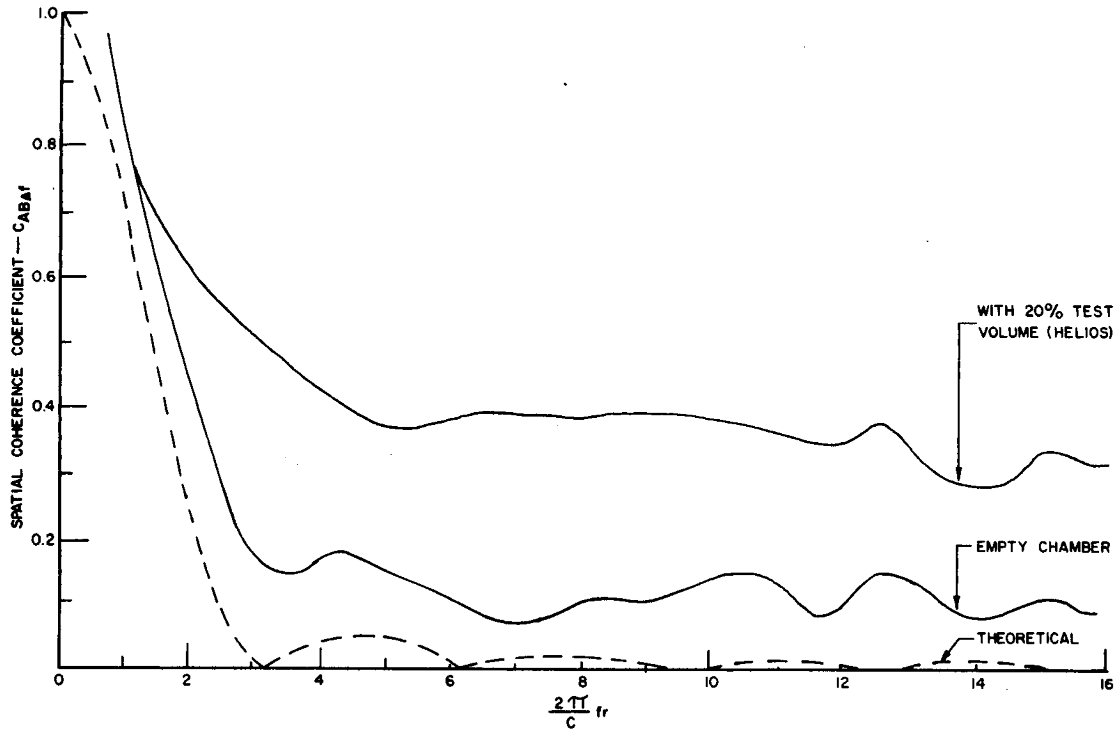

FIGURE 4. SPATIAL COHERENCE BETWEEN MEASURED 1/3 OCTAVE SOUND PRESSURES COMPARED TO THEORETICAL COHERENCE BETWEEN NARROW BAND DIFFUSE SOUND PRESSURES (f = CENTER FREQUENCY, r = DISTANCE BETWEEN POINTS, c = SPEED OF SOUND.)

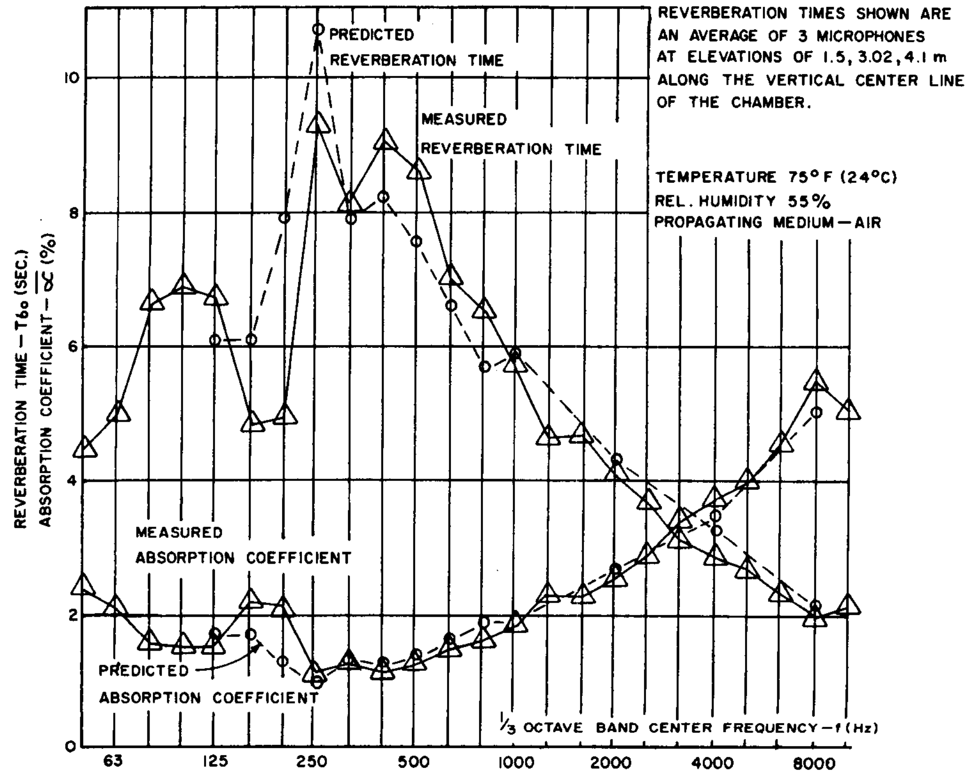

FIGURE 5. PREDICTED AND MEASURED REVERBERATION TIMES AND ABSORPTION COEFFICIENTS-EMPTY CHAMBER

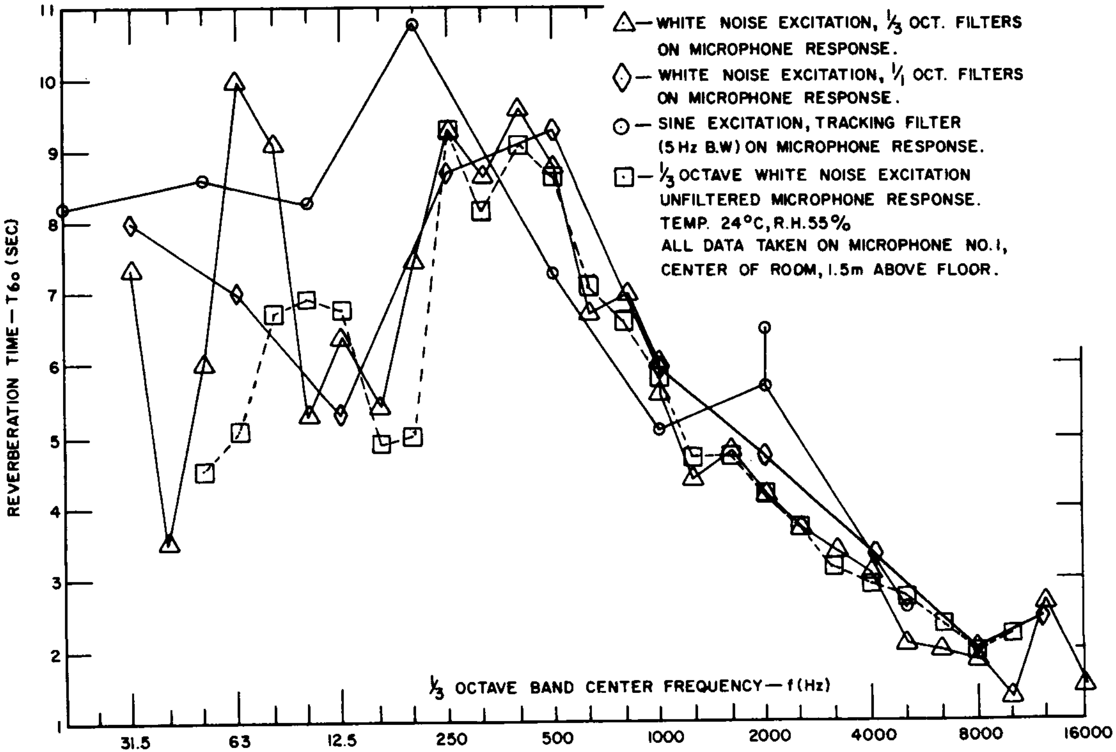

FIGURE 6. REVERBERATION TIMES USING VARIOUS TYPES OF EXCITATION — EMPTY CHAMBER

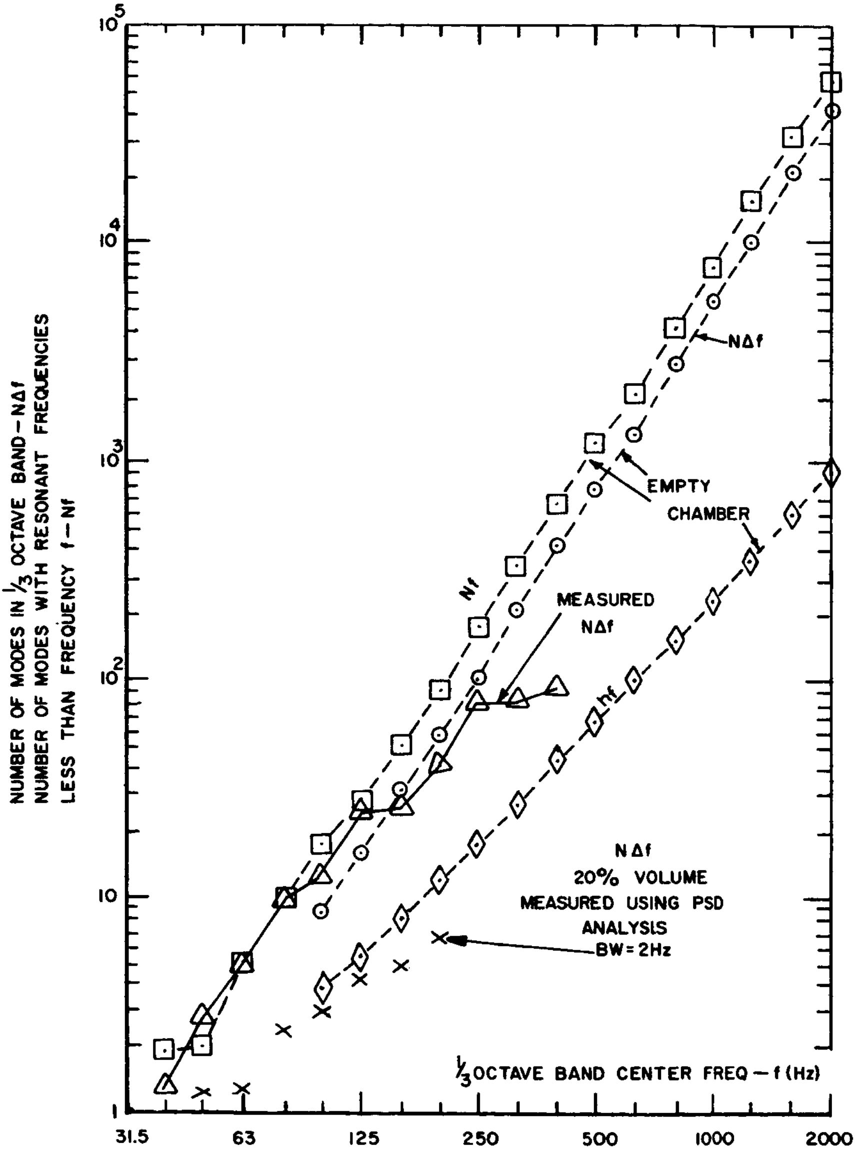

FIGURE 7. PREDICTED MODAL DENSITY (nf), CUMULATIVE RESONANCE FREQUENCY (Nf), AND PREDICTED AND MEASURED MODE COUNTS (NΔf)

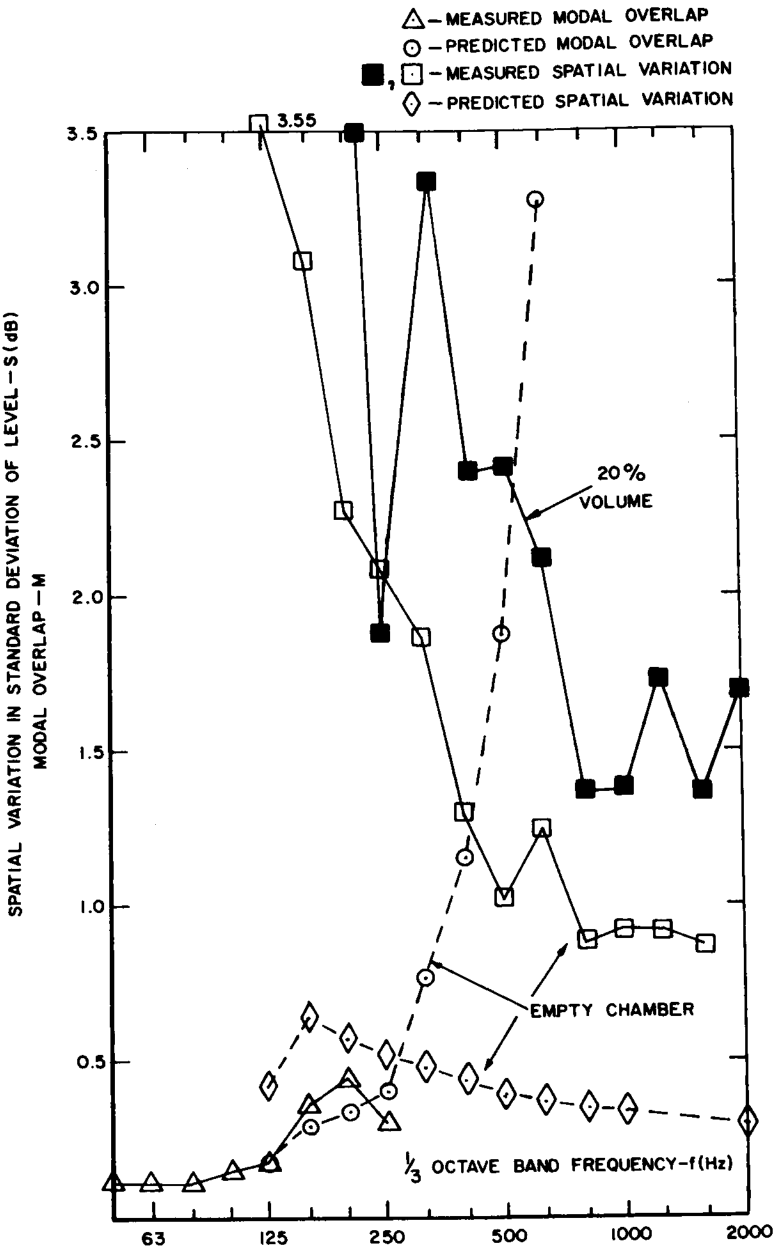

FIGURE 8. PREDICTED AND MEASURED MODAL OVERLAP AND SPATIAL VARIATION–EMPTY CHAMBER AND 20% VOLUME

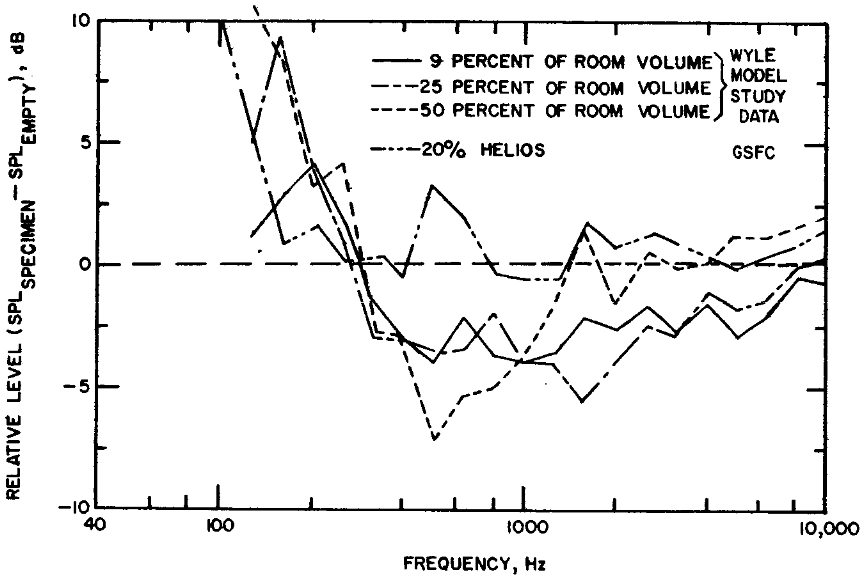

FIGURE 9. SOUND PRESSURE LEVELS MEASURED IN A REVERBERATION ROOM WITH VARIOUS SPECIMEN SIZES RELATIVE TO SOUND PRESSURE LEVELS MEASURED IN THE EMPTY ROOM.